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A LOADING SYSTEM FOR CREEP TESTING UNDER
CONSTANT COMPRESSIVE STRESS

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16. Abstract A critical analysis is made of layouts assuring constant stress under creep, pointing out the difficulties of using these layouts for compressive testing. A new mechanical system to maintain the stress during creep testing is proposed. Constant stress is achieved by a definite motion of the load. The proposed system can be used in tensile or compressive testing. Practical measurements reveal that the stress constancy error is not greater than $\pm 1.5\%$ up to relative deformation of 30%.					
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A LOADING SYSTEM FOR CREEP TESTING UNDER
CONSTANT COMPRESSIVE STRESS

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Introduction

Among the techniques of applying stress during creep testing, uniaxial pressure occupies a significant position. The simple geometry of the specimens preordains this application of stress for the testing of difficult-to-work materials or materials which can be easily damaged by modifications to the desired shape (monocrystals). Leaving aside those materials which are subject to pressure during practical application, creep testing under pressure can also be used to identify the mechanisms which control creep: on the basis of the difference in rates of creep under tension and under pressure, for example, a judgment is made as to the equilibrium concentration of vacancies [1]. For a physical interpretation of the results, it is necessary to maintain constant stress during the test. Whereas the cross section of the specimen diminishes during tensile testing and therefore it is necessary to also reduce the force acting on the specimen, the situation is the complete reverse in the case of compressive testing. In view of the danger of buckling, it is further necessary to employ specimens with a smaller ratio of length to diameter in compressive testing. The use of a creep layout for tensile testing and for testing under pressure at constant stress therefore requires

/599*

*Numbers in the margin indicate pagination in the foreign text.

certain modifications. The present paper proposes an adaptation of the creep layout of the Institute of Physical Metallurgy of the Czechoslovakian Academy of Sciences (UFM ChSAV) [2] for testing under pressure at constant stress.

A Survey of Equipment for Maintaining Constant Stress

A whole variety of systems have been designed for maintaining constant stress during creep testing, based on either electronic or mechanical control of the magnitude of the loading force. The use of electronic systems to produce a constant compressive stress is fundamentally identical to that for constant compressive stress, and therefore we shall not be concerned with such systems hereafter: the disadvantage of these is the fact that they are dependent on the energy source.

/600

Mechanical systems can be classified into six groups, depending on whether the operation makes use of:

1. hydrostatic lifting force,
2. spring pressure,
3. cams,
4. the action of added weights,
5. cranks,
6. forced motion of weights.

Re 1. A hydrostatic lift force acts on the weight, which is dependent on the degree of immersion of the weight in the liquid (Fig. 1). By using a weight of special shape [3], it is possible to achieve a constant tensile stress. This principle cannot be used in compression testing.



Fig. 1. Diagram of a lift-force weighting arrangement as per Andrade [3].



Fig. 2. Diagram of a spring loading arrangement.

Re 2. The force acting on the specimen is produced by a prestressed spring (Fig. 2). The spring tension diminishes as the specimen is plastically deformed. By selecting an appropriate spring stiffness, it is possible to achieve nearly constant tensile stress in a particular deformation range [4]. This principle is also inapplicable for compression testing.

Re 3. Cam systems, originally proposed by Andrade [5], are the most often used to maintain constant stress under creep. The shape of the cam is chosen such that the length of the arm at which the load is applied decreases with the angle of turn of the cam. In order to maintain constant stress under the provision of constant volume of specimen, the length of the arm of load must obey the law (cf. Fig. 3):

$$r = \frac{r_0 l_0}{l_0 \pm r_1 \theta}, \quad (1)$$

where l_0 is the initial length of the specimen, r_0 is the initial length of the arm of load, r_1 is the length of the arm on the side of the specimen (independent of the position of the cam) and θ is the angle of turn of the cam. The plus sign is used

for tension, the minus for compression. The profile of the lever on the side of the load is found by solving the Clairaut differential equation [6]. In parametric form, the profile of the cam is given by the relations:

$$\begin{aligned} x_{1,2} &= \frac{r_0 l_0 [(l_0 \pm r_1 \operatorname{arctg} p) \pm r_1 p]}{\sqrt{1 + p^2 (l_0 \pm r_1 \operatorname{arctg} p)^2}}, \\ y_{1,2} &= \frac{r_0 l_0 [-p(l_0 \pm r_1 \operatorname{arctg} p) \pm r_1]}{\sqrt{1 + p^2 (l_0 \pm r_1 \operatorname{arctg} p)^2}}, \end{aligned} \quad (2)$$

where p is a parameter that can vary from $-\infty$ to ∞ . The plus sign, this time, signifies compression, the minus, tension. The curves produced by equation (2) are symmetrical about the x axis. In practical terms, this means that it is only possible to produce a single profile, which is rotated through 180° about the x axis when the direction of the stress is changed. For the dimensions used at the UFM ChSAV, i.e., $r_0 = 700$ mm, $r_1 = 70$ mm, and the initially assumed nominal length of the specimen $l_0 = 15$ mm, we find that the load at the start of the compression test would have to be suspended on the cam at height $(700 \cdot 70)/15 = 3270$ mm. Therefore, for dimensional reasons, implementation of the Andrade cam is not suitable for the creep layout of the UFM ChSAV. This design is further complicated by the fact that considerable radial displacement of the specimen would occur in the oven of the creep layout during the process of deformation. It is also possible to reduce the dimensions of the cam by employing a multistaged loading system [7]. However, application of this technique would require a new design for the supporting frame of the layout.

A different version of cams has been used at the UFM ChSAV to achieve constant tensile stress [2]. In this version, a convex cam is secured to the end of a lever, which is in contact with a straight link, by means of which a loading force is

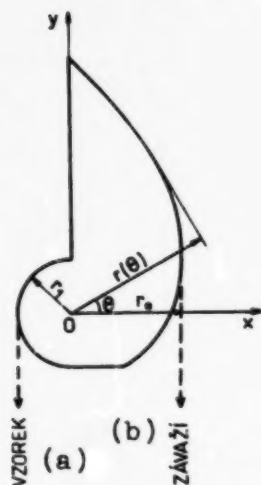


Fig. 3. Diagram of a cam loading arrangement as per Andrade [5]. Key: a - specimen; b - load.

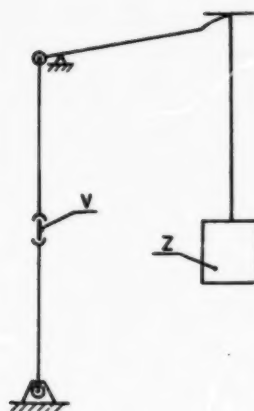


Fig. 4. Diagram of a layout after Hostinsky and Cadek [2].

applied (Fig. 4). To maintain constant compressive stress, the cam would have to be concave in shape. The straight link would then have to be replaced by a rolling element, on which the load should be suspended. The profile of the cam is given by solving the differential equation (assuming a perpendicular tension system):

$$(r_0 + x - yy')(l_0 + r_1 \sin \arctg y') = r_0 l_0. \quad (3)$$

For low values of the angle of turn of the lever, we can solve this equation analytically, obtaining:

$$y = \frac{l_0}{r_1} x + \frac{r_0 l_0}{r_1} \ln \left[\frac{r_0}{r_0 + x} \right]. \quad (4)$$

Inserting the above-mentioned values for r_0 , r_1 and l_0 , we discover that the required profile would be extremely small, and also the actual movement of the rolling element with the load

would be difficult in view of the friction. A further shortcoming of all cam systems is the fact that the shape of the cam is dependent on the initial length of the specimen.

Re 4. A simple modification of a cam system enabling nearly constant stress has been designed by Fullman, et al., [8]. According to this design, the center of gravity of a lever weighing W is situated at distance R beneath a supporting point (this can be achieved, e.g., by the shape of an extra movable load). A nearly constant compressive stress is achieved when:

$$RW = \frac{Pr_0}{l_0}, \quad (5)$$

where l_0 is the length of the specimen at the moment when the lever is in the horizontal position, and P is the weight of the main load. For compression testing, it is necessary to situate the center of gravity of the lever above the supporting point (cf. Fig. 5). The layouts of the UFM ChSAV have been designed for P values up to 70 kg, which with the above-mentioned values of r_1 , r_0 and l_0 results in excessively high values of the product RW , ($= 228.6$ kgm).

Romashkin and Shestopalov have employed a version of the Fullman principle to achieve constant tensile stress [9]. In their layout, an additional lever of rectangular shape with arms of length K and L is connected to the suspended load. At the end of the perpendicular arm of length L is secured an auxiliary load PZ weighing W (cf. Fig. 6). Nearly constant stress is achieved, since:

$$LW = \frac{r_1 PK^2}{r_0 l_0}. \quad (6)$$

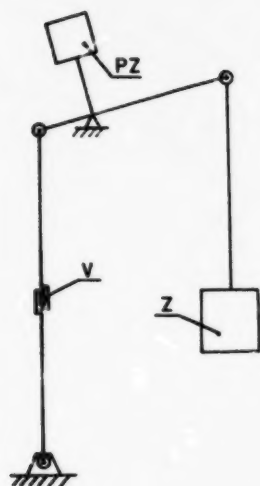


Fig. 5. Diagram of the layout of Fullman, et al., [8].

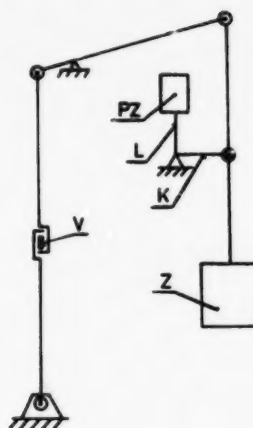


Fig. 6. Diagram of the layout of Romashkin and Shestopalov [9].

As long as the length of the horizontal arm K is shorter than 221 mm, the values of the product LW will be more favorable than indicated by equation (5). The decrease in the length of the arm K is of course limited by the required deformation value. The advantage of being able to use a smaller auxiliary load on a shorter arm than that in the Fullman layout is, however, offset by the necessity of employing a more complicated structural layout. It must be emphasized that both these layouts provide only approximately constant stress. The deviation from constant stress increases with the inclination of the lever θ and is given by the relation:

$$\frac{\Delta\sigma}{\sigma_0} = \frac{r_1}{b} \operatorname{tg} \theta \left(1 - \cos \theta - \frac{r_1}{b} \sin \theta \right), \quad (7)$$

where σ_0 and l_0 are the stress and the length of the specimen at the instant when the lever is in the horizontal position.

Re 5. The layout using a crank [10] is based on the identical physical principle (cf. Fig. 7). Nearly constant stress will be achieved if the arm of the load and the arm of the specimen constitute an angle γ such that:

$$\operatorname{tg}(180^\circ - \gamma) = \frac{l_0}{l},$$

where l_0 is, again, the length of the specimen at the instant when the arm of the specimen is in the horizontal position. The orientation of the angle γ is opposite for tensile and for compressive stress. The deviation from constant stress is the same as in the Fullman layout. Application of this principle in the layout of the UFM ChSAV again results in dimensional problems: the crank on the side of the load would have to be a minimum of 3900 mm long.

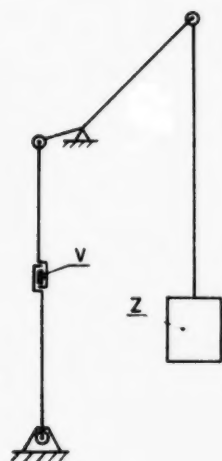


Fig. 7. Diagram of the layout of Sellars and Quarell [10].

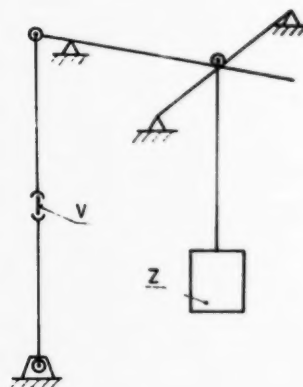


Fig. 8. Diagram of the layout of Hoffmann [11].

Re 6. According to the kinematic analysis of Andrade and Chalmers [5], constant stress under loading by a lever will be achieved if the point at which the load acts on the lever moves

along a line having a directrix $\operatorname{tg} \alpha = l_0/r_1$, where l_0 is the length of the specimen in the horizontal position of the lever. (The orientations of the angle α are opposite in tensile and compressive stress.) In the practical implementation of this principle, the motion of the loading point along the desired path must be delimited by an accessory arrangement. The weight of the load is, of course, distributed between the lever and the accessory arrangement. Taking this fact into account, the point at which the load acts on the lever must travel along a complicated surface curve. In the horizontal position of the lever, this curve will have a directrix $\operatorname{tg} \alpha' = 2l_0/r_1$. Hoffmann [11], instead, proposed to replace this curve with a straight line and to have a guide plate of the same slope as the accessory arrangement. The structural implementation consists in suspending a load from a ball bearing, moving in the valley between the lever and the guide plate (Fig. 8).

/604

The layout proposed by Hoffmann functions with precision only if the lever is in the horizontal position. The uncertainty of the layout depends on the initial length of the specimen, in addition to the slope of the lever. With a suitable choice of the ratio of the length of the specimen to the length of the lever arm, it is possible to achieve very slight deviations from constant stress, even when the specimen is quite elongated. A shortcoming of the method, however, is the necessity of constructing a special layout to limit the motion of the load. If such layout is provided with an adjustment of the slope of the guide plate, the system can be used with different initial nominal lengths of specimen, either for compression or for tension.

/605

It follows from the above survey that the layout presented under points 3-6 can theoretically be used in the design of a layout for creep deformation under constant compressive stress.

So far as we know, principles 3 [12, 13], 4 [9] and 5 [10] have been used in practice for compression testing. Electronic control of the loading force has been used in [14]. It is further evident from the analysis that adaptation of these systems for the creep layout of the UFM ChSAV is attended by difficulties, primarily of a dimensional nature.

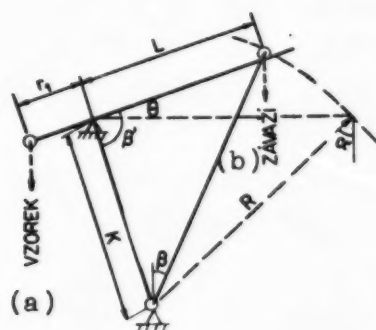


Fig. 9. Kinematic diagram of the proposed design.
Key: a - specimen; b - load.

Design of the Layout

The proposed design of modifications of a lever-type loading system proceeds from the same principles as the design of the Hoffmann layout [11]. The difference consists in the fact that the complex curve along which the load must move is replaced by a circle of large radius. As in the Hoffmann layout, the load is suspended by a ball bearing, which is simultaneously secured to one end of a rod. The other end of the rod is rotationally restrained in the base or in another fixed component of the creep layout (cf. Fig. 9). The position of this rotational point is chosen such that the directrix of the rod in the horizontal position of the lever is $\text{tg}(90^\circ - \alpha')$. The end of the lever on the side of the load is provided with a level

plate, along which the ball bearing of the load suspension can travel as an inclined plane. The point at which the load will act on the lever will be given by the intersection of the instantaneous position of the lever and the circle with radius equaling the length of the rod with center at the rotational point. In the horizontal position of the lever, this circle has a tangent of directrix $\text{tg } \alpha'$, i.e., the same as the ideal complex curve. With increasing deviation of the lever from the horizontal position, just as in the Hoffmann straight-line layout the deviation from the ideal curve and thus the inaccuracy of the arrangement will increase. However, the proposed layout has the advantage that it can operate with positive or negative deviations of the lever position, whereas the Hoffmann layout is limited either to positive or negative deviations, depending on whether we are producing tensile or compressive stresses (the directrix of the lever and that of the guide plate must have opposite sign). This means that, in order to achieve the same level of deformation, the proposed design requires half the deviation of the lever, which results in substantially greater accuracy. A further advantage is the simplicity of the structural layout: uprights carrying the guide plate are not needed. Compared to the cam systems, the proposed design has the advantage of greater universality. A special cam need not be used for each nominal length of specimen: it is enough to change the position of the rotational point, or the length of the rod.

/606

The relative deviation of the stress as a function of the inclination of the lever in the presented layout is given by the relation (assuming a constant volume of specimen and constant perpendicularity of the tension system):

$$\frac{\Delta\sigma}{\sigma} = \frac{L \sin \beta}{L_0 \sin (\Theta + \beta) \cos \Theta} \left(1 + \frac{r_1 \sin \Theta}{l_0} \right) - 1,$$

where β is the instantaneous angle between the supporting rod and the vertical. L and L_0 signify the length of the lever on the side of the load in any given position and in the horizontal position, and can be determined by means of the equation:

$$L = \frac{K + \sqrt{R^2(1 + \tan^2 \beta') - K^2 \tan^2 \beta'}}{\sqrt{1 + \tan^2 \beta'}}$$

where R is the length of the rod, K is the distance between the supporting point of the lever and the rotational point of the rod, and β' is the angle between the line connecting these points and the lever. The meaning of the symbols used is obvious from Fig. 9.

Implementation of the Layout and Verification of Its Operation

The proposed principle of maintaining constant compressive stress made it possible to employ the creep layout of the UFM ChSAV for compressive stress testing with minimal alterations. A detailed description of this creep layout is given in [2]. The modification for constant compressive stress (cf. Fig. 10) requires a lever provided with a plate along which the ball bearing of the load suspension can travel, and a furnishing of the load suspension with such bearing, which is at the same time connected to the upper end of a bracing rod. The bracing rod is formed by a Dural tube with threaded adapters, which delimit its required length. To the base plate of the creep layout, in addition, is secured a bracket with openings for rotational restraint of the lower end of the rod. Furthermore, it is necessary to modify the jaws in which the specimen is restrained so that the motion of the lever in the same direction corresponds to compressive, instead of tensile stress. The upper jaw has the shape of a hook, on which the bottom surface of the specimen rests. The lower jaw is designed in the shape of a dismountable frame, which acts on the upper surface of the specimen.

/607

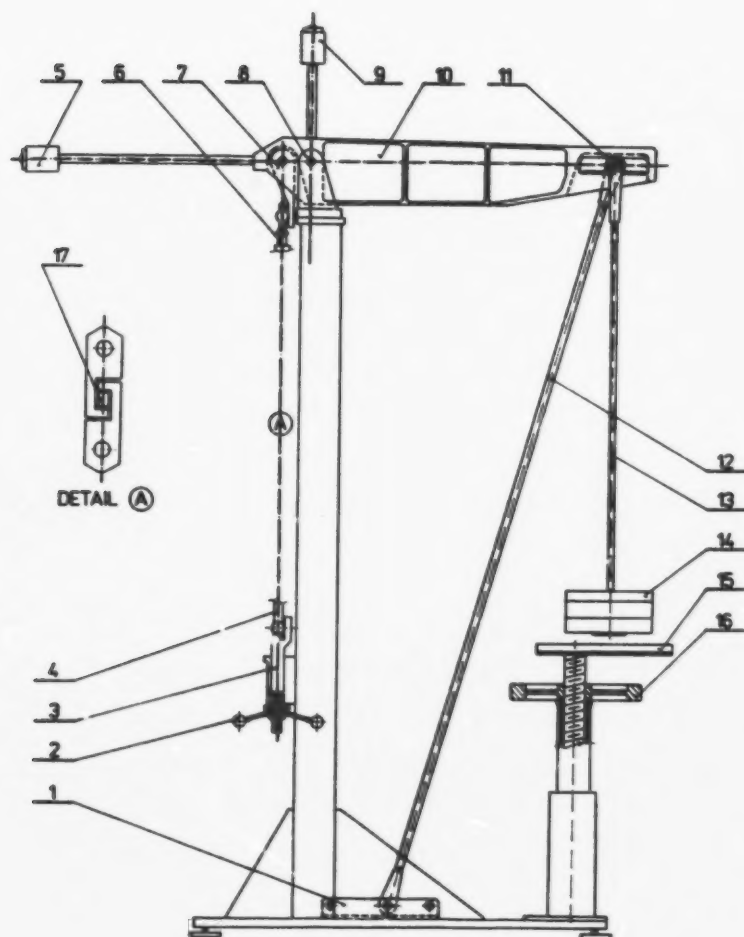


Fig. 10. Structural layout of the compression modification of the UFM ChSAV creep apparatus: 1 - bracket for rotational restraint of the lower end of the bracing rod; 2 - adjustment mechanism; 3 - bottom suspension; 4 - bottom tie-rod; 5 - counterweight; 6 - upper tie-rod; 7 - upper suspension; 8 - fulcrum of lever; 9 - counterweight; 10 - lever; 11 - bearing; 12 - bracing rod; 13 - load suspension; 14 - load; 15 - carrying plate of loading mechanism; 16 - flywheel of loading mechanism; 17 - specimen in schematically diagrammed jaws.

All modifications were made in terms of testing of specimens with initial nominal length of 15, 12 and 9.5 mm. These

nominal lengths correspond to rod lengths of $R = 2022$, 1979 and 1938 mm. In view of the fact that the remainder of the creep apparatus is unchanged, the lever transmission when using the aforesaid rod lengths is not $1:10$, as in the original version for constant tensile stress, but $1:9.287$, $1:1.928$ and $1:10.15$ for the corresponding nominal lengths.

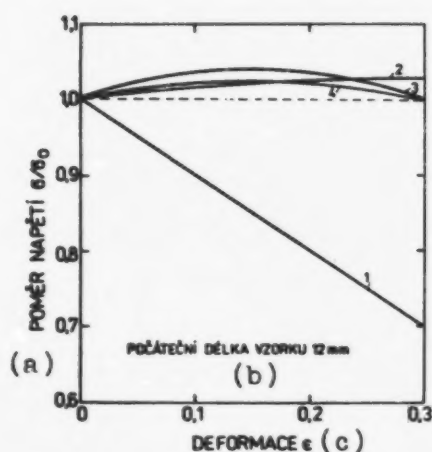


Fig. 11. Relative stress as function of deformation for various loading systems: 1 - constant loading; 2 - Hoffmann [11]; 3 - Fullman, et al., [8]; 4 - system of the authors. Key: a - stress ratio; b - initial length of specimen 12 mm; c - deformation.

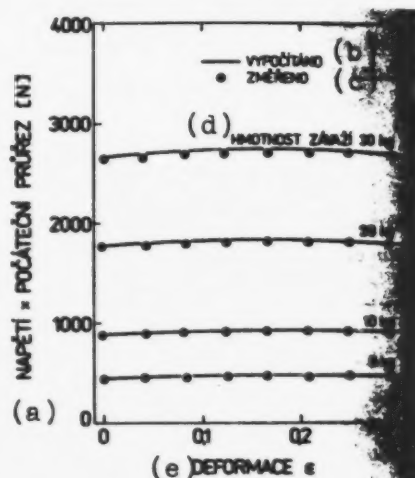


Fig. 12. Comparison of obtained and computed relations between relative stress and deformation. Initial length of specimen 12 mm. Key: a - stress $\times$ initial cross section [N]; b - calculated; c - measured; d - weight of load; e - deformation.

The theoretically calculated curves of the stress for different starting nominal lengths of specimen in the case of the modifications made in the UFM apparatus are shown in Fig. 11. The difference between the highest and the lowest stress is not more than 3%, even up to deformation of 30%. The overall stress curve is better than that of the Hoffmann layout under similar geometrical conditions.

The proper functioning of the proposed design was verified by measuring the force transmitted by the lever to the specimen. This measurement was taken by dynamometric clips in the range up to 200 and up to 400 kp. The deformation of the clips was ascertained by microscope with precision of 10^{-6} m (force measurement error 0.07% max). The corresponding stress was calculated from information as to the position of the upper jaw, determined by a clockwork indicator with precision of 0.01 mm, under assumption of constant volume of the imaginary test specimen. The measurement results for various nominal lengths and various weights of the load are shown in Fig. 12. It is clear from the diagram that the measured variation of forces corresponds quite well to the anticipated. Preliminary creep testing further demonstrated that the operation of the apparatus is continuous in the entire deformation process.

Conclusion

The paper proposes a system for maintaining nearly constant stress during creep testing. This system employs forced motion of a load on a lever. The movement of the load is bounded by means of a rotationally restrained rod. In principle, this system is similar to the Hoffmann layout for maintaining a constant stress, but has the advantage of structural simplicity and greater accuracy. In contrast with the cam systems, it has the advantage of greater universality. It is particularly suitable for small nominal length of specimen, in which case the required dimensions of the cams are unacceptably large. The proposed system has been implemented in the creep apparatus of the UFM ChSAV. Static measurement of the force transmission and continuous operation in dynamic testing have demonstrated the validity of the prototype.

/609

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